

Research Paper

Experimental comparison and optimization guidance of R1233zd(E) as a drop-in replacement to R245fa for organic Rankine cycle application

Jingye Yang, Ziyang Sun, Binbin Yu, Jiangping Chen*

Institute of Refrigeration and Cryogenics, Shanghai Jiao Tong University, Shanghai 200240, PR China



HIGHLIGHTS

- A design of extensive operating conditions in experimental comparison.
- R1233zd(E) was proven as an appropriate substitute to R245fa in current research.
- Volume ratio dependent curves of expander efficiency well predict the expander.
- Non-dimensionless indicator for comparison of alternative in heat exchanger.
- An optimization guidance was proposed for future work.

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ABSTRACT

The organic Rankine cycle is capable of converting the low-temperature waste heat into electricity. The commonly used working fluid R245fa will be phased out in the near future because of the significant impact to climate change. In that case, a new refrigerant R1233zd(E) with extremely low GWP is proposed as an environmental friendly substitute in this paper. The investigation is processed with four steps: firstly, a thermodynamic analysis was carried out for prior prediction of the applicability of R1233zd(E) as an alternative to R245fa; secondly, an experimental comparison between two refrigerants was implemented under a design of extensive operating conditions; Afterwards, experimental results were presented. Differences in expansion and evaporation procedure based on three non-dimensionless indicators were analyzed; Eventually, a multi-objective optimization guidance involved with aforementioned indicators was proposed. Comparing the maximum cycle thermal efficiency, R1233zd(E) leads to approximately 3.8% higher than R245fa. Comparing the maximum output electrical power, R1233zd(E) leads to 4.5% better than R245fa. R1233zd(E) is proven as an appropriate alternative to R245fa based on current study. Prediction precision of the volume ratio dependent curves of filling factor and isentropic effectiveness are within 1.2% and 2.7%, which can be used to model a certain expander in optimization procedure.

1. Introduction

Increasing consumption of fossil fuels and carbon dioxide emission have raised the concern about the sustainable development. The organic Rankine cycle is proposed as a viable combined heat and power (CHP) technology for low-temperature waste heat recovery from engines [1,2], which in turn improves the fuel energy efficiency. Working fluid selection is one of the primary considerations of organic Rankine cycle that depends on the heat resource characteristics and operating conditions [3].

Several screening criteria for working fluid selection in waste heat recovery applications have been proposed in literatures. Blake et al. [4]

presented the first test methodology for thermal stability of working fluids applied to more than 100 organic fluids. Based on the same methodology, a specific investigation of working fluids utilized in organic Rankine cycles was carried out by Invernizzi [5]. R245fa was assumed as an optimal choice for low-grade temperature heat source. Different configurations of organic Rankine cycle were economically assessed by Imran et al. [6] and the results indicated that R245fa has the lowest specific investment for low-grade heat resource. Lecompte et al. [7] developed a thermo-economic design methodology for the optimization of organic Rankine cycle based on the specific investment cost (SIC). Part load behavior under time-varying operating conditions was taken into consideration and the results of 1-year simulation

* Corresponding author.

E-mail address: jpchen_sjtu@163.com (J. Chen).

Nomenclature			
A	heat transfer area, m ²	HFO	hydrofluoroolefins
c _p	specific heat capacity, J/kg K	ICE	internal Combustion Engine
f	fanning friction number, –	LCOE	levelized cost of electricity
h	heat transfer coefficient, W/m ² K	ODP	Ozone Depletion Potential
j	Colburn factor, –	PHE	Plate Heat exchanger
$\dot{m}$	mass flow rate, kg/s	SIC	specific investment of cost
Nu	Nusselt number, –	SF	size factor
P	pressure, kPa	VOC	Volatile Organic Compounds
Pr	Prandtl number, –		
Q	heat rate, W		
Re	Reynolds Number, –		
r _v	built-in volume ratio, –		
T	temperature, °C		
Acronyms		Subscripts	
CHP	combined heat and power	cp	compressor
GWP	Global Warming Potential	ev	evaporator
HFC	hydrofluorocarbon	exp	expander
		ex	exhaust
		in	inlet
		is	isentropic
		pp	pump
		rot	rotational
		ref	refrigerant
		su	supply

revealed that the net power varies drastically over a year. He et al. [8] developed a thermodynamic simulation model to analyze five fluids at different evaporation temperature and condensation temperature. R236fa showed the highest thermal efficiency as working fluid applied in organic Rankine cycle system to improve fuel economy.

HFCs (Hydrofluorocarbons) with high GWP (Global Warming Potential) [9] will be phased out in the future because of significant impact to the environment. Similar thermodynamic properties are proposed as the first standard for the selection of drop-in replacements. Criteria of alternative molecules should consider environmental impact such as GWP, ODP, VOC (Volatile Organic Compounds) and safety indicators such as non-toxicity and non-flammability. Manufacturability including cost and materials compatibility is important as well. Gary Zyhowski and Andrew Brown [10] implemented a comparison of HFOs with HFCs in all aspects and concluded that HDR-14 allowed to achieve a higher thermal efficiency than R245fa in organic Rankine cycle. Van Long Le et al. [11] investigated a supercritical organic Rankine cycle using R1234yf, R1234ze(E) and R152 as working fluids. R152 always leads to the highest optimal efficiency both in basic and regenerative organic Rankine cycles (ORCs). The maximum net power was obtained with R1234ze(E) in basic configuration. Moles et al. [12] theoretically compared R1233zd(E) and R1336mzz(Z) as alternatives to R245fa. Evaporating temperature was set between 97 °C and 147 °C while condensing temperature was set between 27 °C and 77 °C. Superheating ranged from 0 °C to 50 °C. Results showed that R1233zd(E) and R1336mzz(Z) led to 10.6% and 17% higher thermal efficiency than R245fa, respectively.

Currently, limited experimental investigations on alternative refrigerants have been reported in literature because of limited availability of HFOs on the market. Datla and Brasz [13] in 2014 were the first to carry on an experimental study with a 75-kW basic ORC using radial inflow turbine. They showed that R1233zd(E) led to 8.7% higher net cycle efficiency and more output electricity than R245fa. Matthais Welzl et al. [14] presented a comprehensive evaluation of organic Rankine cycle combining nucleate boiling heat transfer coefficient (HTC) with output electricity. R245fa showed higher HTC and performed better than R1233zd(E) at the same saturation temperature due to higher pressure and density. Sebastian Eyerer et al. [15] reported that R1233zd(E) performed approximately 7% higher thermal efficiency but with 12% lower gross power output.

Applicability of R1233zd(E) as a drop-in replacement to R245fa in ORC system has been analyzed in previous literatures. Nevertheless,

different focus point of investigation results in distinct design of operating conditions. Thus, experimental results can be diversified.

The aim of this paper is not only to prove the potential of R1233zd(E) as working fluid to recover low-temperature waste heat but also to propose an optimization guidance for better taking advantage of new refrigerant in current system, which hasn't been mentioned before. The design of operating conditions differs from previous researches. Volume ratio dependent curve-fittings of non-dimensionless expander efficiency were implemented, which can be used in the optimization procedure. Moreover, an objective equation was first-time proposed to compare the refrigerants in a certain heat exchanger. Non-dimensionless Colburn factor and Fanning friction number are both taken into consideration.

This paper is processed with four sections: first of all, a fundamental thermodynamic comparison was implemented to give a prior prediction of R1233zd(E) as a feasible alternative to R245fa in ORC; secondly, a drop-in experimental testing of R1233zd(E) was conducted in a baseline ORC test rig designed for R245fa; afterwards, the experimental results were presented; meanwhile, further discussions in expansion and evaporation procedure were carried out simultaneously; last but not the least, a conceptual optimization guidance was proposed for better taking R1233zd(E) in current system as a reference method.

2. Properties comparison of R1233zd(E) and R245fa

2.1. Thermo-physical properties

Hydrofluorocarbons (HFCs) were considered as greenhouse gases according to Kyoto Protocol [16] in 1997 because of the significant impact to climate change. R245fa has a GWP of approximately 1030. According to the European Parliament [17], HFCs with a GWP higher than 150 have to be banned in domestic refrigerators by 2015. R1233zd(E) is suggested as a drop-in replacement to R245fa considering the similar thermodynamic properties with extremely low GWP. Table 1 summarizes the thermo-physical properties of R1233zd(E) and R245fa [15].

2.2. Thermodynamic analysis

The thermodynamic analysis is aimed to give a prior prediction of the new refrigerant R1233zd(E) as an applicable alternative to R245fa. The primary thermodynamic comparison under certain operating conditions between two refrigerants is also prepared for the lateral

Table 1
Thermo-physical properties of R1233zd(E) and R245fa [15].

Parameters	R1233zd(E)	R245fa
Chemical formula	$\text{CF}_3\text{CH}=\text{CHCF}_3$	$\text{CF}_3\text{CH}_2\text{CF}_2$
P_c (kPa)	3570	3650
T_c (°C)	165.6	154.01
Boiling point (°C)	17.97	14.81
Latent heat at boiling point (kJ/kg)	195.52	196.23
MM (kg/kmol)	130.5	134
Slope	Dry	Dry
ODP	0.00034	0
GWP _{100yr}	7	1030
Atmospheric life time	7.7	0.07
Flammability	Non-flammable	Non-flammable
ASHARE Std 34 safety class	A1	B1
Decomposition temperature (°C)	> 250	n.a.

Table 2
Numeric values for parameters.

Parameters	Numeric value
Evaporating temperature (°C)	70, 75, 80, 85, 90
Condensing temperature (°C)	30
Superheating (°C)	5
Subcooling (°C)	5
Pump/expander efficiency (–)	0.7

Table 3
Governing equations.

Component	Formula
Pump	$W_{pp} = \dot{M}_{ref} \Delta h = \dot{M}_{ref} v_{in} \Delta p / \eta_{pp}$
Heat exchanger	$Q = \dot{M}_{ref} \Delta h$
Expander	$\eta_{is,exp} = \Delta h / \Delta h_{is}$ $W_{exp} = \dot{M}_{ref} \eta_{is,exp} \Delta h_{is}$
Cycle	$\eta_{cycle} = (W_{exp} - W_{pp}) / Q_{ev}$

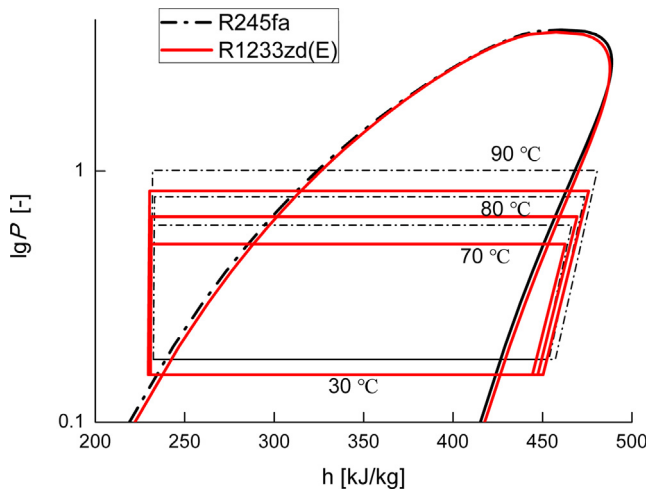


Fig. 1. $\lg P$ - h diagram of R1233zd(E) and R245fa in ORC.

experimental investigation. Additionally, formulas for calculation of cycle performance are presented in this section.

Table 2 shows the numeric value for each parameter, where evaporating temperature is chosen as the state variable and rest of them are kept as constants. Pressure drop and heat transfer loss are neglected during the thermodynamic calculation. Governing equations are summarized in Table 3. Where W_{pp} and W_{exp} are referred to the pump consumption and expansion work; $\eta_{is,exp}$ and η_{pp} are expander isentropic

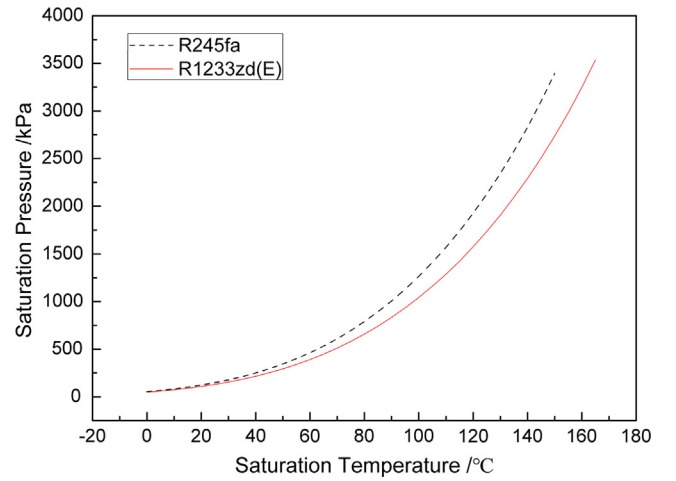


Fig. 2. Saturation pressure curves of R1233zd(E) and R245fa.

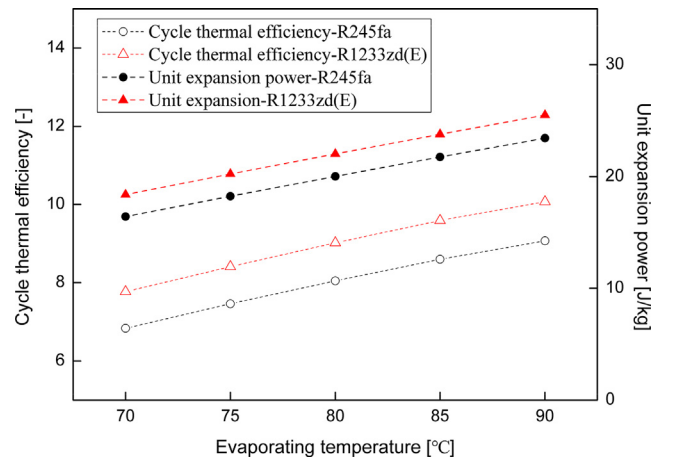


Fig. 3. Comparison of cycle thermal efficiency and unit expansion power between R1233zd(E) and R245fa.

efficiency and pump efficiency, respectively; $\dot{M}_{ref}$ is referred to the mass flow rate; v_{in} is the specific inlet volume flow rate; Q_{ev} is the heat transfer rate in the evaporator. Thermal properties of two refrigerants are obtained from the Equation of state (EOS) based software REFPROP-NIST 9.1.

Fig. 1 shows the $\lg P$ - h diagram of R1233zd(E) and R245fa in the organic Rankine cycle under three different evaporating temperatures. For the same operating temperatures, the corresponding evaporating pressures of R1233zd(E) are lower than R245fa, which can be explained by the saturation curves in Fig. 2. In other words, for the same given operating pressure, R1233zd(E) can achieve higher evaporating temperature than R245fa. Fig. 3 presents evolutions of cycle thermal efficiency and unit expansion power to evaporating temperature. Higher evaporating temperature contributes to the better cycle thermal efficiency and more unit expansion power for both refrigerants. R1233zd(E) always leads to higher cycle thermal efficiency and generates more unit expansion power than R245fa in current operating range.

In conclusion, the new low-GWP refrigerant R1233zd(E) is theoretically proven as a potential substitute to R245fa for low-temperature organic Rankine cycle application.

3. Experimental apparatus

3.1. Test bench description

The schematic of conventional organic Rankine cycle system is

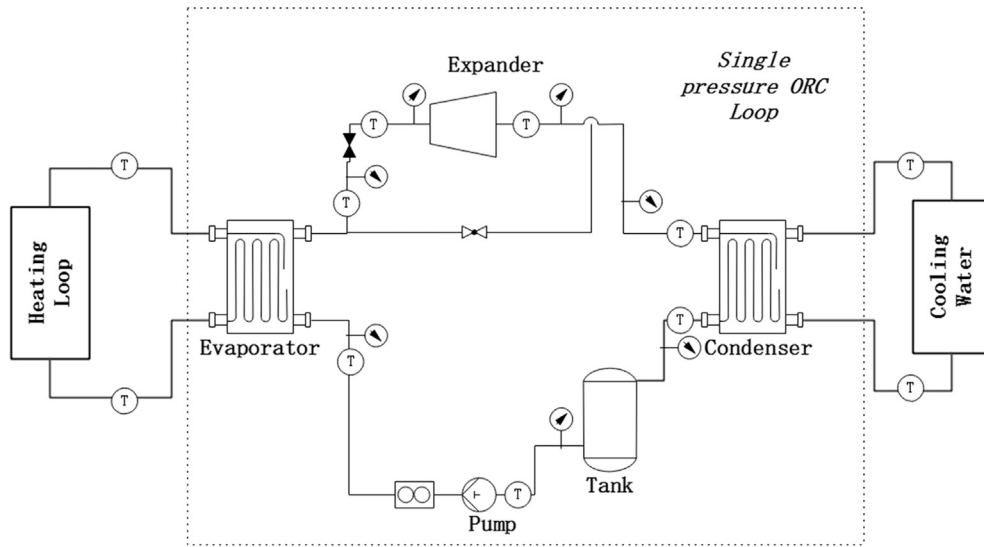


Fig. 4. Test rig of organic Rankine cycle.

Table 4
Boundary conditions.

Loop	Inlet temperature [°C]	Flow rate [m ³ /h]
Heating loop	100	3.9
Cooling loop	27.5	8

presented in Fig. 4. Four main components consist of the basic ORC system including feed pump, evaporator, condenser and expander. The temperature and mass flow rate of the low-grade heat source are imitated by an electrical oil heater. The cooling loop consists of a water circuit and a cooling tower. Boundary conditions of organic Rankine cycle are summarized in Table 4.

Fig. 5a shows a plunger diagram pump, which is considered appropriate for small scale ORCs since high pressure can be achieved with small mass flow rate. The maximum pressure of the liquid pump is 3000 kPa. A 1.1 kW 3-phase 380 V motor was magnetically coupled to this pump.

The grazed plate heat exchanger was chosen in current study because of better performance than shell-and-tube in the heat transfer process [18], as shown in Fig. 5b. An open-drive scroll expander which was modified from a commercially available refrigeration compressor with a swept volume of 86 cm³ had been installed in the ORC test rig and it is shown in Fig. 5c. Suction and discharge valves of the scroll machine were removed in order to operate as an expander. The built-in volume ratio of such scroll machine is around 2.5. Scroll expander is commonly used in small scale cogeneration system due to compact structure with less moving parts, reliability, operational simplicity, lower level of noise and lower production costs [19–20]. A 2 kW 3-phase AC generator was coupled to the modified scroll expander with belt. The loaded side was designed adjustable with a rheostat. The maximum expander rotational speed is constrained by the operating conditions and the limitation of the coupled generator rotational speed (1800 rpm). Pressure and temperature sensors were installed at each inlet and outlet of components. A Coriolis-type mass flowmeter was placed at the outlet of liquid pump. Table 5 summarizes the measuring range and accuracy of each sensor.

3.2. Design of operating conditions

Cycle performance of organic Rankine cycle is a comprehensive result of process parameters. The integral ORC system is treated as a ‘black-box’ with two degrees of freedom ($\dot{M}$ and N_{rot}) in Fig. 6.

Refrigerant is the only input element of the ‘black-box’ that makes difference. Mass flow rates and expander rotational speeds are assigned as control options. There are nine operating conditions in total, which can be classified into two types in Table 6: under the fixed expander rotational speed or under the fixed mass flow rate. In the first part of the table, the mass flow rate is varied for five scroll expander rotational speeds (1150 rpm, 1250 rpm, 1350 rpm, 1450 rpm and 1550 rpm). In the second part of the table, expander rotational speed is varied for four mass flow rates (100 kg/h, 120 kg/h, 140 kg/h, and 160 kg/h) by adjusting the loaded side of the generator. In this case, various couplings of mass flow rate and expander rotational speed are explored.

The adjustable range under each operating condition is different, such method enables the thorough exploration of working points in this ‘black-box’ to find the optimum cycle performance for a given refrigerant. A total of 63 steady-state points for each working fluid have been collected based on the steady-state standard.

3.3. Uncertainties and steady-state standard

Accuracy of each measurement device has been listed in Table 5. The uncertainty analysis is implemented based on the formula developed by Moffat [21]. Uncertainty of dependent variable is calculated by Eq. (1), where R is a function of independent variables and w represents their uncertainties. Based on such method, the calculated accuracy of cycle thermal efficiency, output electrical power and expander efficiency are within 1%, 0.8% and 1.1%, respectively.

$$W_R = \left(\left(\frac{\partial R}{\partial X_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial X_2} w_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial X_n} w_n \right)^2 \right)^{1/2} \quad (1)$$

Measurements are recorded using a digital multi-meter and an armature relay multiplexer. A steady-state is ensured if the fluctuation range of the measured parameters is lower than 1% on each variable. During the measurement period of 10 min, all measured values are logged every second.

4. Results and discussion

This section is conducted with three steps: first of all, system performance indicators including cycle thermal efficiency and output electrical power were presented; afterwards, simultaneous analysis in expander and evaporator was implemented; eventually, a conceptual optimization guidance for better taking advantage of R1233zd(E) was proposed.

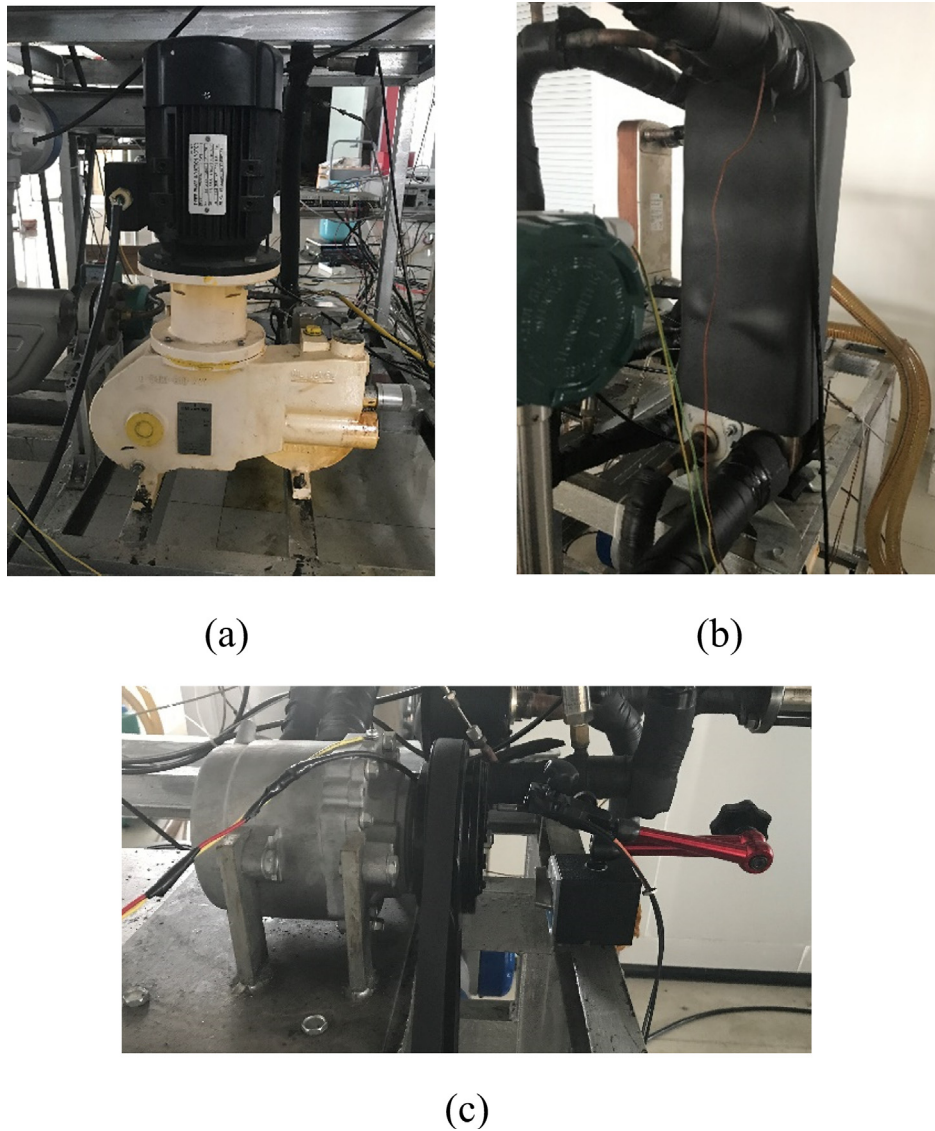


Fig. 5. (a). Plunger diaphragm pump (b). Plate heat exchanger (c). Open-drive scroll expander.

Table 5

Instrumentation used to obtain system measurements.

Measurement	Type	Range	Accuracy
Temperature	RTD 100	− 200 °C to + 300 °C	± 0.5%
Pressure Abs	Electronic pressure transducer (gauge)	0–1000 kPa	± 0.5%
Mass flow rate (working fluid)	Coriolis effect	0–600 kg/h	± 0.5%
Volume flow rate (Cold water)	LWGY-10BIC1	1–10 m ³ /h	± 0.5%
Rotational speed	FT7I-D60B	0–9999 rpm	± 0.3%
Power monitor	DU4-W	0–2500 W	± 0.3%

4.1. Experimental results

4.1.1. System performance

Fig. 7 depicts the evolution of cycle thermal efficiency under each operating condition. In Fig. 7(a), expander rotational speeds were fixed at 1150 rpm, 1250 rpm, 1350 rpm, 1450 rpm and 1550 rpm, respectively. Mass flow rates are adjusted to different values to form the performance curve. Increasing mass flow rate brings up the evaporation

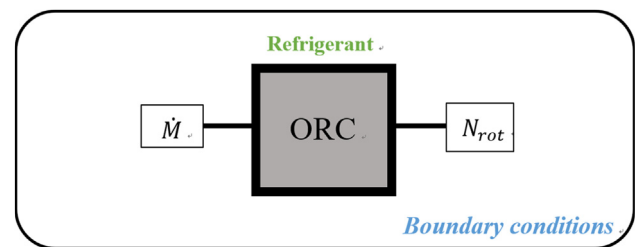


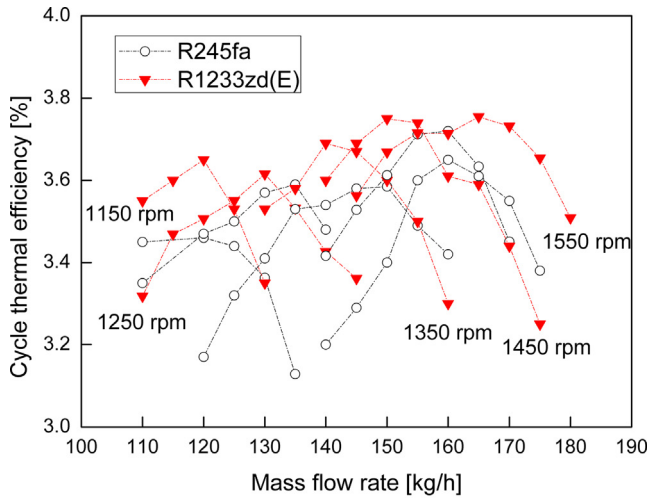
Fig. 6. Concept of black-box.

Table 6

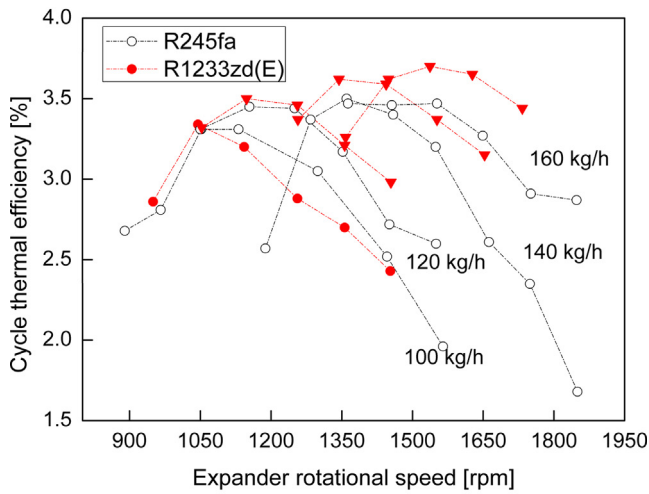
Summary of operating conditions.

Operational parameters	Expander rotational speed	Mass flow rate
Expander rotational speed (rpm)	1150, 1250, 1350, 1450, 1550	Adjustable
Mass flow rate (kg/h)	Adjustable	100, 120, 140, 160

pressure, which contributes to more output electricity. However, the pump consumption and heat rate absorbed in evaporator increase as well. This explains the existence of the peak of efficiency in each curve



(a)



(b)

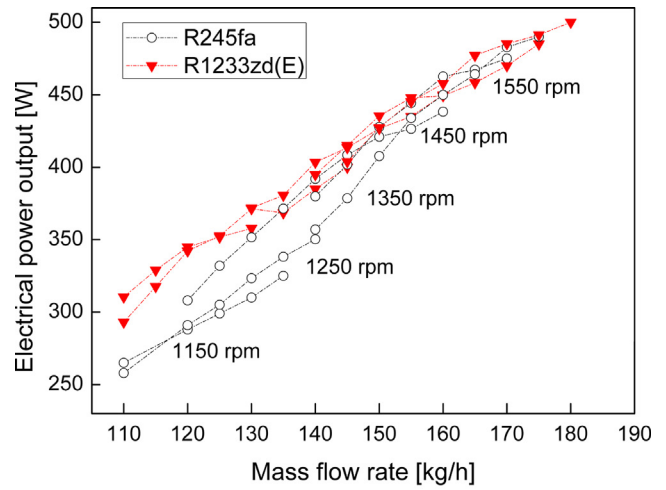
Fig. 7. (a). Evolution of cycle thermal efficiency to mass flow rate (b). Evolution of cycle thermal efficiency to expander rotational speed.

for both refrigerants.

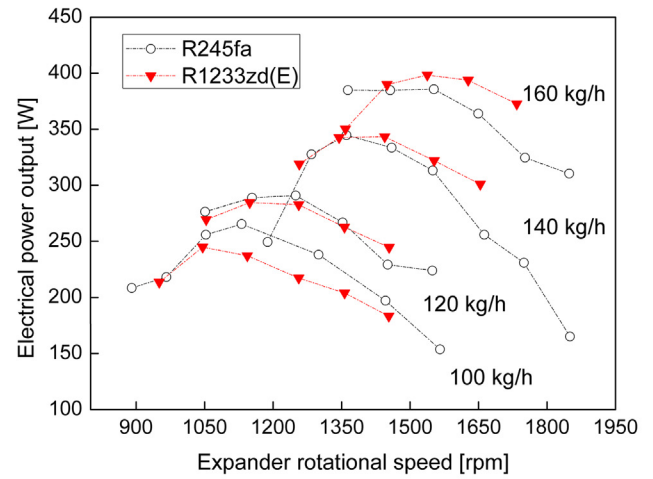
In Fig. 7(b), mass flow rates were fixed at 100 kg/h, 120 kg/h, 140 kg/h and 160 kg/h, respectively. The spread of cycle thermal efficiency is due to different expander rotational speeds. Increasing expander rotational speed improves the internal leakage problem while brings down the evaporation pressure at the same time. Obviously, there also exists a peak of efficiency in each curve.

Fig. 8 describes the evolutions of output electrical power. In Fig. 8(a), under each fixed expander rotational speed, the output electrical power increases with mass flow rates for both refrigerants. This can be explained by the higher evaporation pressure with larger mass flow rate. Another important point is, the adjustable mass flow rate range increases with higher fixed expander rotational speed. Hence, output electricity under higher fixed rotate speed seems always larger than that under the lower one.

The trend of output electrical power in Fig. 8(b) is different from that in Fig. 8(a). Peak of electricity occurs in each curve. Under each fixed mass flow rate, the adjustable expander rotational speed influences the evaporating pressure, internal leakage inside the expander and mechanical friction at the same time. Low expander rotational speed results in severe internal leakage that decreases output electrical power. High expander rotational speed brings down the evaporating pressure and increases the mechanical friction loss. With understanding



(a)



(b)

Fig. 8. (a). Evolution of electrical power output to mass flow rate (b). Evolution of electrical power output to expander rotational speed.

the influence of abovementioned factors, an optimum expander rotational speed is assumed for a fixed mass flow rate

Cycle thermal efficiency is considered as a synthesis performance indicator of ORC system. In that case, peaks of efficiency in each curve are summarized in Fig. 9 for further comparison between two refrigerants. Based on the exploration of all working points in current operating condition range, R1233zd(E) always leads to higher cycle thermal efficiency than R245fa. The results show the maximum cycle thermal efficiency achieved with R1233zd(E) is 3.76% while in case of R245fa is 3.62%. Comparing the maximum cycle thermal efficiency, R1233zd(E) leads to approximately 3.8% better than R245fa. Concerning the maximum difference in cycle thermal efficiency, R1233zd(E) leads to approximately 5% better than R245fa. Therefore, R1233zd(E) is proven as an appropriate drop-in replacement to R245fa in current study.

Another important fact reveals that cycle thermal efficiency is a comprehensive result of two control options. Various couplings of mass flow rate and expander rotational speed may result in different behavior in evaporator and expander, which in turn influence the system performance. On account of this, further discussion should be implemented in expander and evaporator simultaneously.

4.1.2. Expansion performance

Two dimensionless expander efficiency including expander

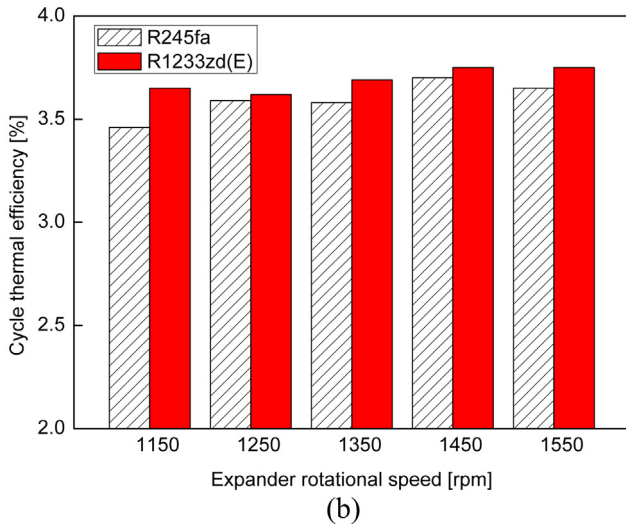
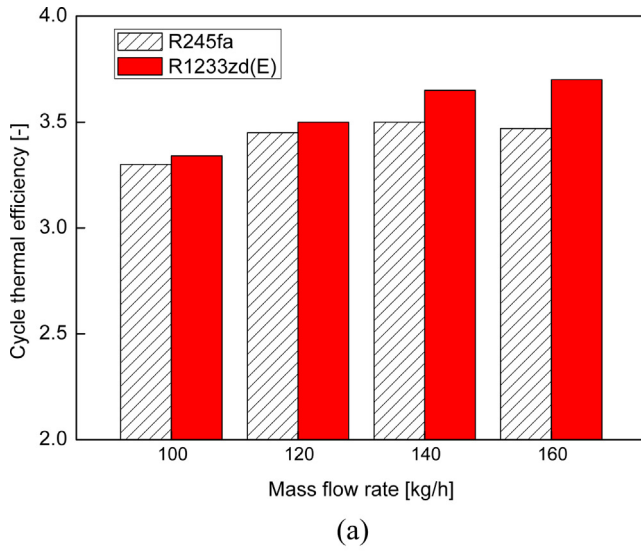


Fig. 9. (a). Peaks of cycle thermal efficiency for four mass flow rates (b). Peaks of cycle thermal efficiency for five expander rotational speeds.

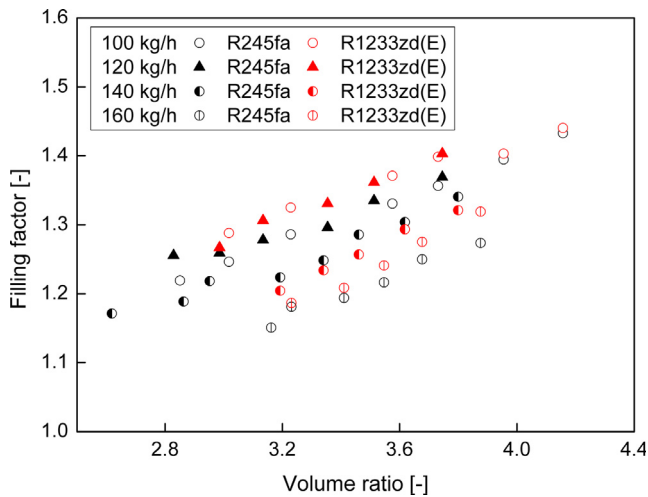


Fig. 10. Evolution of filling factor to expansion volume ratio.

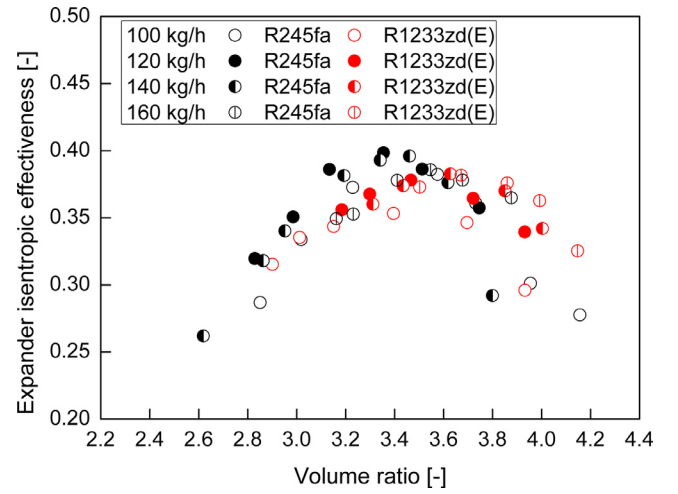


Fig. 11. Evolution of expander isentropic effectiveness to expansion volume ratio.

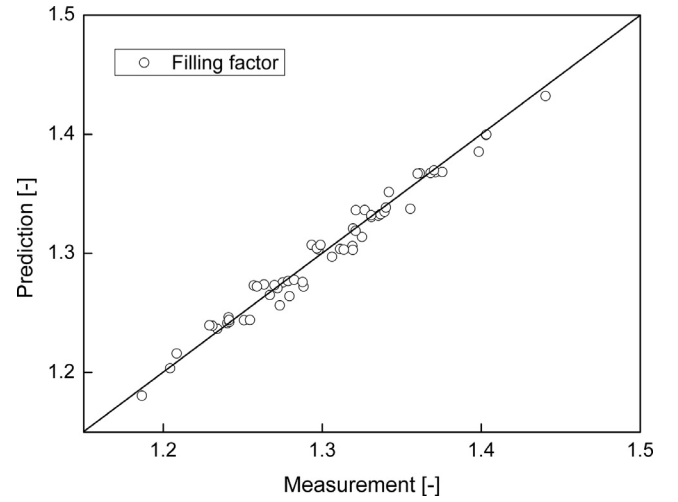


Fig. 12. Prediction precision of filling factor.

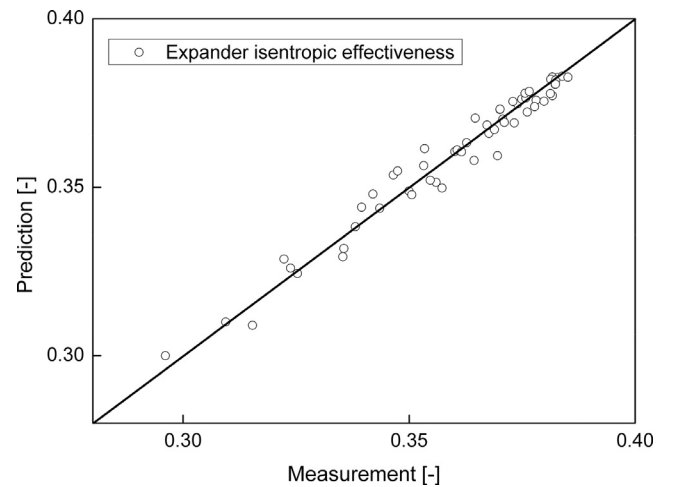


Fig. 13. Prediction precision of expander isentropic effectiveness.

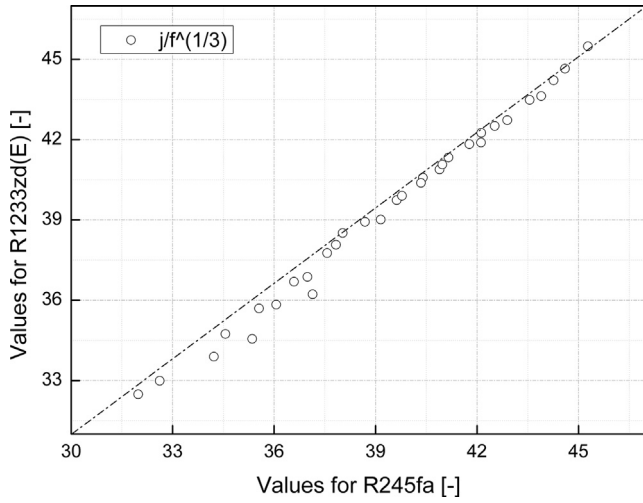


Fig. 14. Comparison of non-dimensional indicator between two refrigerants.

isentropic effectiveness and filling factor can give an important insight into an expander performance. The filling factor shown in Eq. (2) is used to qualify the volumetric performance of scroll expander [22,23]. If the filling factor is greater than unit, the volume flow rate displaced per revolution is higher than the ideal geometrical value.

$$\phi = \frac{M_{ref} v_{su}}{N_{rot} V_{s,exp}} = \frac{M_{ref} v_{su}}{N_{rot} V_{s,cp}/r_v} \quad (2)$$

where M_{ref} is the refrigerant mass flow rate, $V_{s,cp}$ and $V_{s,exp}$ represent swept volume of scroll machine in compressor mode and expander mode, respectively; r_v is the built-in volume ratio and v_{su} is the specific volume flow rate at the inlet of the expander. Lastly, N_{rot} is the expander rotational speed.

In Fig. 10, mass flow rates are respectively fixed to 100 kg/h, 120 kg/h, 140 kg/h and 160 kg/h. The spread of filling factor is due to different expansion volume ratio, which is defined as a ratio of exhaust and supply specific volume of the expander. Higher expansion volume ratio indicates larger pressure difference that contributes to more internal leakage for both refrigerants. Except that, filling factor is influenced by expander rotational speed and working fluid density. Literally, lower density of R1233zd(E) results in the larger filling factor than R245fa under the same expander rotational speed. Increasing expander rotational speed can effectively decrease the internal leakage where

tiny difference in density is negligible.

Expander isentropic effectiveness is defined as a ratio of the output electricity to the ideal expansion power in Eq. (3).

$$\varepsilon_{is} = \frac{P_{scroll}}{M_{ref} (h_{su,exp} - h_{ex,is,exp})} \quad (3)$$

where P_{scroll} is the electrical output power, $h_{su,exp}$ and $h_{ex,is,exp}$ are supply specific enthalpy and isentropic exhaust specific enthalpy of the expander, respectively;

Fig. 11 depicts the evolution of expander isentropic efficiency to expansion volume ratio under the same given operating conditions in Fig. 10. Peak of effectiveness occurs in each curve. R1233zd(E) shows slightly lower expander isentropic effectiveness than R245fa. Expander isentropic effectiveness is a comprehensive result of filling factor and expansion volume ratio. The deviation between the operating volume ratio and the built-in volume ratio will decrease the expansion performance drastically. Lower effectiveness of R1233zd(E) can be explained by both higher volume ratio than R245fa and smaller density that contributes to more internal leakage.

Methodology for well prediction of expander performance in system simulation is expected, which is aimed for further optimization of R1233zd(E). Based on the observation of relationships between the non-dimensional expander efficiencies and the operating volume ratio, curve-fittings of filling factor and isentropic effectiveness for R1233zd(E) were implemented. All working points were involved and results are presented in Eq. (4) and (5). Figs. 12 and 13 show the deviation between the prediction results and the measured results. The prediction precisions for filling factor and isentropic effectiveness are within 1.2% and 2.7%, respectively. Such two non-dimensional curves enable to model a certain expander for a given refrigerant in a simplified way.

$$\phi_{ff,R1233zd(E)} = 1.687 + 0.01134 \cdot V_{ratio} - 0.0003151 N_{rot} \quad (4)$$

$$\varepsilon_{is,R1233zd(E)} = -3.713 + 1.403 \cdot V_{ratio} + 2.468 \cdot \phi_{ff} - 0.1469 \cdot V_{ratio}^2 - 0.2627 \cdot V_{ratio} \cdot \phi_{ff} - 0.6109 \cdot \phi_{ff}^2 \quad (5)$$

4.1.3. Heat transfer performance in evaporator

In this section, the comparison of heat transfer process in evaporator between R1233zd(E) and R245fa was conducted. Expander rotational speeds were fixed at 1150 rpm, 1250 rpm, 1350 rpm, 1450 rpm and 1550 rpm respectively. Mass flow rates were adjusted to different values. It's well known that increasing mass flow rate enhances the heat

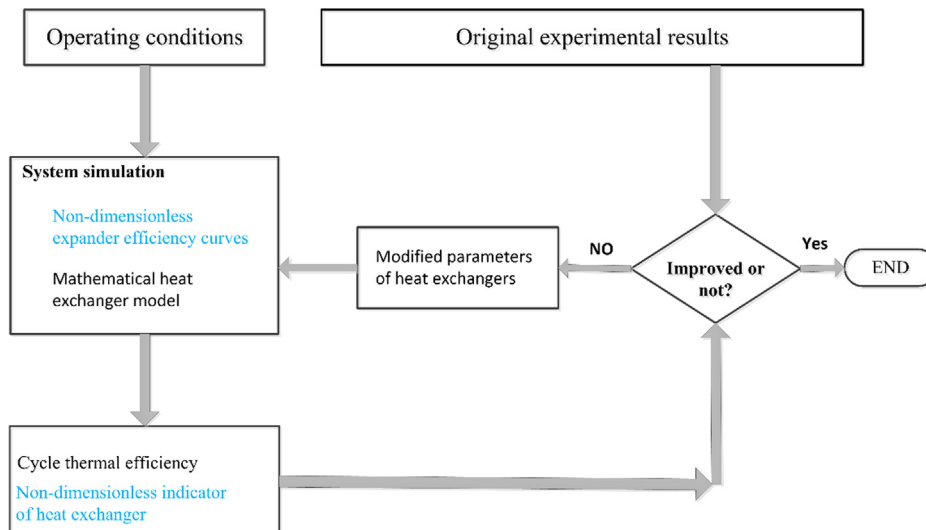


Fig. 15. Conceptual optimization procedure.

transfer while increases pressure drop at the same time. In that case, a non-dimensionless objective equation [24] for comparison of substitute to the reference refrigerant (e.g. R245fa in this study) in a certain evaporator was proposed in this paper.

$$\xi = \frac{\frac{j}{j_{ref}}}{\left(\frac{f}{f_{ref}}\right)^{1/3}} \quad (6)$$

where j is referred to the Colburn factor that is related to heat transfer process; f is the Fanning friction number that is related to the pressure drop. Empirical formulas for plate type heat exchanger were developed by Chisholm and Wanniarachchi (1991), which are presented in Eqs. (7) and (8); Where β is the chevron angle (e.g. 60° in this paper) and formulas are fit for $3000 < Re < 10^5$.

$$j = 0.72 \cdot \left(\frac{\beta}{30}\right)^{0.66} \cdot Re^{0.59} \quad (7)$$

$$f = 0.8 \cdot \left(\frac{\beta}{30}\right)^{3.6} \cdot Re^{-0.25} \quad (8)$$

Lumped parameter method was used for calculation of refrigerant thermal properties. Average values of j and f were obtained for both working fluids. Fig. 14 presents the calculated results of working points in current operating conditions. Results of the non-dimensionless objective equation range from 0.99 to 1.01, which reveals that R1233zd(E) shows very similar heat transfer performance to R245fa. The advantages of such non-dimensionless indicator have twofold:

- (1) For synthesis evaluation of new substitute in a certain heat exchanger compared to the baseline refrigerant.
- (2) For optimization of heat transfer performance to better take advantage of a given refrigerant. The non-dimensionless objective function can be used to compare the refrigerant performance in the modified heat exchanger to the baseline one.

4.2. Optimization guidance

System performance relies on the matching between components. For this reason, selection of appropriate heat exchangers to match with a certain expander for a given refrigerant is important. To better take advantage of R1233zd(E), an optimization guidance involved with three non-dimensionless indicators was proposed. Fig. 15 shows the conceptual schematic of the optimization procedure. Input parameters are assigned to the operating conditions and modified geometric parameters of heat exchangers. Aforementioned two expander efficiency curves for R1233zd(E) should be integrated with mathematical heat exchanger models to constitute the system simulation. Back and forth validation of the mathematical system simulation based on experiments is demanded to improve the accuracy of optimization guidance. Multi-objective optimization of R1233zd(E) in current system can be realized with an optimization algorithm, where cycle thermal efficiency and the non-dimensionless indicator of heat exchanger are chosen as objective functions. Such optimization guidance along with the three non-dimensionless factors are recommended as references for future work of substitute investigation.

Validation of the mathematical system simulation and further optimization procedure based on experimental results are complicated and out of scope in this paper, which will be presented in the next step of research.

5. Conclusions

An experimental comparison between R1233zd(E) and R245fa in ORC system was implemented in this paper. Furthermore, a conceptual optimization guidance for better taking advantage of R1233zd(E) was

proposed. Several conclusions can be drawn as follows:

- (1) R1233zd(E) was proven as an applicable drop-in replacement to R245fa in organic Rankine cycle from the thermodynamic point.
- (2) After the exploration of working points in current operating conditions, results show that the maximum cycle thermal efficiency achieved with R1233zd(E) is 3.76% while in case of R245fa is 3.62%. Comparing the maximum cycle thermal efficiency, R1233zd(E) leads to approximately 3.8% better than R245fa. Comparing the maximum gross output electrical power, R1233zd(E) generates 4.5% more electricity than R245fa since higher mass flow rate can be achieved under the same expander rotational speed.
- (3) R1233zd(E) indicates higher expansion volume ratio and more internal leakage inside the expander than R245fa due to lower density, which result in the lower expander isentropic effectiveness of R1233zd(E). Increasing the expander rotational speed can decrease the internal leakage. In order to improve the isentropic effectiveness of R1233zd(E), the designed volume ratio of the expander should be modified to adapt to the higher operating volume ratio in future work.
- (4) Two non-dimensionless expander efficiency curves were fitted to model a certain expander in a simplified way. Prediction precision for filling factor and isentropic efficiency are within 1.2% and 2.7%, respectively.
- (5) A non-dimensionless indicator of heat exchanger for comparison between two refrigerants was first-time proposed. Both the characteristics of heat transfer and pressure drop were taken into consideration. R1233zd(E) shows very similar performance to R245fa in current heat exchanger based on such objective equation.
- (6) An optimization guidance for choosing appropriate heat exchangers to match with a certain expander for a given refrigerant was proposed. Mathematical heat exchanger model should be integrated with aforementioned expander efficiency curves to consist of the system simulation. The multi-objective optimization can be realized based on an optimization algorithm, where both cycle thermal efficiency and non-dimensionless indicator of heat exchanger are chosen as objective functions.

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Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at <http://dx.doi.org/10.1016/j.applthermaleng.2018.05.105>.

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